

BEARING CUP ROTATIONAL LOCK ASSEMBLY

FIELD OF THE INVENTION

[0001] The present invention relates to bearing assemblies and more
5 particularly to a bearing cup for a bearing assembly.

BACKGROUND OF THE INVENTION

[0002] Typically, rolling element bearings are used in rotating
machinery designs (e.g. liquid rocket engine turbopumps) to provide radial and
10 axial support of a rotating shaft assembly. These rolling element bearings are
often a single ball bearing or a pair of preloaded angular contact ball bearings.
The bearings are typically mounted to the rotating shaft and provide axial and
radial positional control of the shaft.

[0003] One method of obtaining adequate radial and axial load carrying
15 capabilities with some amount of damping is to use a combination of preloaded
angular contact ball bearings and a hydrodynamic bearing. Yet many designs,
including turbopumps for rocket engine applications, typically incorporate a
balance piston to control the axial position of the shaft at various operating
speeds. The balance piston utilizes the controlled pressures in a fluid flow circuit
20 to provide axial thrust of the shaft while balancing the loads applied to the turbine.
However, the angular contact ball bearings only control the axial position of the
rotor during start-up and shut-down conditions. At operating speed, the axial
position of the shaft is controlled by the balance piston, as noted above. The
amount of axial movement of the shaft is considerable and the design requires
25 that the outer race of the bearing assembly be slidably fitted in the bearing
support housing.

[0004] However, a well performing hydrodynamic bearing will center
the shaft at operating speeds, potentially eliminating contact of the bearing outer
race with the bearing support housing. In the absence of contact, the bearing
30 outer race will spin and potentially gall as the race intermittently contacts the
bearing housing. This spinning and galling of the bearing outer race can lead to
part failure.

[0005] Accordingly, it would be highly desirable to provide a mechanism for preventing this failure by implementing a bearing cup rotational locking assembly that allows for significant shaft axial travel while preventing rotational movement of the outer race of the bearings.

5

SUMMARY OF THE INVENTION

[0006] A bearing cup apparatus for use with a bearing is provided. In one preferred embodiment, the apparatus includes a ring and a shoulder extending radially inward from an inner surface of the ring. At least one tang extends axially outward from a first side of the ring. At least one tooth extends axially outward from a second side of the ring opposite the first side. At least one slot is formed in the second side. The apparatus prevents the outer race of a conventional rolling element bearing from rotating while allowing the bearing to move in an axial direction. The apparatus can be readily used with either a single rolling element bearing, a set of two bearing assemblies, or any number of closely spaced bearing assemblies. The advantage of the apparatus is that it eliminates spinning of the bearing assembly outer race. The apparatus also prevents frictional sliding between a bearing assembly outer race and a preload spring. Frictional sliding imposed on a bearing outer race can induce galling and subsequently lead to part failure.

[0007] Further areas of applicability of the present invention will become apparent from the detailed description provided hereinafter. It should be understood that the detailed description and specific examples are intended for purposes of illustration only and are not intended to limit the scope of the invention.

BRIEF DESCRIPTION OF THE DRAWINGS

[0008] The present invention will become more fully understood from the detailed description and the accompanying drawings, wherein:

5 [0009] Figure 1 is a partial cross sectional view of an exemplary turbopump having a bearing cup rotational lock assembly constructed according to the principles of the present invention

[0010] Figure 2 is an exploded perspective view of the bearing cup rotational lock assembly;

10 [0011] Figure 3 is a perspective view of the bearing cup rotational lock assembly;

[0012] Figure 4 is a front perspective view of a bearing cup of the bearing cup rotational lock assembly constructed according to the principles of the present invention;

15 [0013] Figure 5 is a back perspective view of the bearing cup of Figure 4; and

[0014] Figure 6 is a cross sectional view of the bearing cup rotational lock assembly indicated by the box 6-6 shown in Figure 1.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

20 [0015] The following description of the preferred embodiment(s) is merely exemplary in nature and is in no way intended to limit the invention, its application, or uses.

[0016] Referring to Figure 1, a bearing cup rotational lock assembly (bearing assembly) 10 constructed according to the principles of the present invention is shown mounted in an exemplary turbopump 12. It is to be understood, however, that the bearing assembly 10 may be employed in numerous other mechanical devices having one or more bearings, including engines, turbines, or rotating shafts.

30 [0017] The turbopump 12 generally includes a housing 14, a damper seal 16, a shaft 18, and a rotating component 20. The bearing assembly 10 is mounted within the housing 14 and supports the shaft 18 for rotation. The damper seal 16 provides rotordynamic damping and radial support to the shaft 18. The rotating component 20 is mounted on the shaft 18

[0018] Turning to Figures 2 and 3, the bearing assembly 10 is illustrated in greater detail. In one preferred embodiment, the bearing assembly 10 includes a first bearing 30, a first bearing cup 32, a second bearing 34, a second bearing cup 36, a spring 38, and a shim 40. The first and second bearings 30, 34 and the first and second bearing cups 32, 36 are substantially identical. Accordingly, only the first bearing 30 and first bearing cup 32 will be described in detail, it being understood that the detailed description applies equally to the second bearing 34 and second bearing cup 36, respectively. In this regard, the various components of the second bearing 34 and second bearing cup 36 will be designated with the number of the component corresponding to the first bearing 30 and the first bearing cup 32 followed by a " ' " symbol.

[0019] The first bearing 30 is illustrated as a preloaded angular contact ball bearing as is known in the art. However, various other rolling element bearing assemblies may be employed. The first bearing 30 includes an outer race 42, an inner race 44, a bearing cage 45, and a plurality of balls 48 (two of which are visible). The inner race 44 is rotatably supported by the plurality of balls 48 within the outer race 42. The outer and inner races 42, 44 are generally ring shaped.

[0020] Referring now to Figure 4, the first bearing cup 32 is generally ring shaped and defines an axis indicated by line A-A in Figures 4 and 5. The first bearing cup 32 includes a front side 50, a rear side 52, an outer surface 54, and an inner surface 56. A pair of tangs 58 extend out from the front side 50 in the direction of axis A-A. While only two tangs 58 are illustrated, it is to be understood that as many or as few as one tang may be employed. The tangs 58 fit within a portion of the housing 14 and prevent rotation of the first bearing 30 while allowing axial movement therein, as will be described in greater detail below. A chamfered edge 60 is formed along the front side 50 and transitions the front side 50 to the outer surface 54. The chamfered edge 60 aids in inserting the bearing cup 32 into the housing 14 during assembly of the bearing assembly 10. The chamfered edge 60 also prevents galling from axial travel during operation. A rounded edge could be substituted for the chamfered edge.

[0021] As best seen in Figure 5, the first bearing cup 32 further includes a plurality of teeth 62 extending out from the rear side 52 in the direction of axis A-A. A plurality of slots 64 are formed between each of the plurality of teeth 62 on the rear side 52. The plurality of slots 64 are sized to receive the plurality of teeth 62 from the second bearing cup 36 as will be described in greater detail below. While in the particular example provided six teeth 62 and six slots 64 are illustrated, it is to be understood that any number of teeth and slots may be employed. Moreover, the first bearing cup 32 can be used singly by itself in which case no teeth or slots are required.

[0022] A shoulder 66 is formed on the inner surface 56 adjacent the rear side 52 and extends radially inward. The shoulder 66 is sized to accommodate the spring 38 as will be described below. Moreover, the bearing cup 32 can have lubricated surfaces to reduce sliding friction. The bearing cup surfaces in contact with the outer race 42 are not lubricated in the preferred configuration.

[0023] The bearing cup 32 is sized to fit over the outer race 42 of the first bearing 30. The inner diameter of the inner surface 56 of the first bearing cup 32 is smaller than the outer diameter of the outer race 42 such that the bearing cup 32 is press fitted onto the bearing 30 thereby creating an interference fit between the two. In the preferred embodiment, the bearing cup 32 is constructed from a high strength steel, although various other materials may be employed.

[0024] Returning to Figure 2, in the particular example provided the spring 38 is a cylindrical beam spring. Compression of the spring 38 creates a wave shape and preloads the outer races 42 of the first and second bearings 30, 34 in the direction of axis A-A. Alternatively, other biasing members may be used for the spring 38 such as a Belleville spring.

[0025] Turning now to Figure 6, the interrelationship of the various components of the bearing assembly 10 will be described in greater detail. The first and second bearings 30, 34 are mounted onto the shaft 18 between the rotating component 20 and a shaft shim 70. The shaft 18 extends through the inner races 44, 44' of the first and second bearings 30, 34 and is supported for rotation therein. The inner races 44, 44' are rotatably and axially fixed to the

shaft 18. The first and second bearings 30, 34 are spaced apart from one another by the shim 40 mounted therebetween.

[0026] As noted above, the first and second bearing cups 32, 36 are press fitted onto the first and second bearings 30, 34, respectively. In this regard, the inner surfaces 56, 56' have an interference fit with the outer races 42, 42'. Preferably, the bearing cups 32, 36 are installed on the outer races 42, 42' by heating the bearing cups 32, 36 and chilling the outer races 42, 42' such that the bearing cups 32, 36 expand and the outer races 42, 42' contract. The interference fit that results is designed such that the load imparted on the outer races 42, 42' result in a negligible change in the size of the bearing raceway of the outer races 42, 42'. In addition, the press fit is sized such that the hoop stresses in the bearing cups 32, 36 remain within desired limits and a satisfactory amount of fit is maintained during all operating conditions. The first and second bearing cups 32, 36 are fixed to the first and second bearings 30, 34 such that the rear sides 52, 52' (Figures 2 and 4) of the first and second bearing cups 32, 36 face one another.

[0027] As best seen in Figure 3, the first and second bearing cups 32, 36 engage one another. The teeth 62 of the first bearing cup 32 fit within the slots 64' of the second bearing cup 36 while the teeth 62' of the second bearing cup 36 fit within the slots 64 of the first bearing cup 32. In the preferred embodiment, the slots 64, 64' are wider than the teeth 62, 62' thereby creating a gap, indicated by reference numeral 72. This allows the bearing cups 32, 36 to rotate slightly before locking each other from further rotation.

[0028] Returning to Figure 6, it can be seen that there is a further gap in the axial direction between the teeth 62, 62' and the slots 64, 64' such that the bearing cups 32, 36 may move in the axial direction. The spring 38 is mounted between the first and second bearing cups 32, 36 and is enclosed by the teeth 62, 62', and by the shoulders 66, 66'. The shoulders 66, 66' in turn engage the outer races 42, 42' of the bearings 30, 34. The shoulders 66, 66' further act to ensure that a proper axial fit with the bearings 30, 34 is achieved when the bearing cups 32, 36 are mounted thereon.

[0029] After assembly, the spring 38 exerts a force in the direction of axis A-A and urges the outer races 42, 42' away from each other. This in turn

urges the plurality of balls 48 against the inner races 44, 44' thereby preloading the bearings 30, 34. In addition, the shoulders 66, 66' prevent frictional sliding between the spring 38 and the outer races 42, 42'.

5 **[0030]** Pockets 74 (one of which is shown) are formed within the housing 14 for receiving the tangs 58 of the first bearing cup 32. In the particular example provided, the pockets 74 are illustrated as being formed in part by the housing 14 and the damper seal 16. However, the pockets 74 may alternatively be formed entirely by the housing 14 (not shown) or entirely by the damper seal 16 (not shown). The pockets 74 have a depth, indicated by reference numeral 76,
10 that is greater than the length of the tangs 58, indicated by reference numeral 78. In this way, movement of the bearing assembly 10 in the direction of axis A-A will never lead to the tangs 58 from escaping the pockets 74 or the tangs 58 from bottoming in the pockets 74. In addition, the width and thickness of the tangs 58 are designed such that the stresses induced in the tangs 58 are within desired
15 limits for all loading conditions.

[0031] The bearing cups 32, 36 can significantly extend the operational life of a turbopump 12. At the start of an engine (not shown), the preloaded bearing assembly 10 will provide radial support to the shaft 18. As the engine (not shown) transitions to operating speeds, a balance piston (not shown) will
20 control the position of the shaft 18 in the direction of axis A-A and the bearing assembly 10 will slide to accommodate the change in axial position. The radial loads between the housing 14 and the bearing assembly 10 may be completely eliminated. In the event that environmentally induced torque on the outer races 42, 42' is greater than resisting frictional load between the bearing cups 32, 36
25 and the housing 14, the bearing cup tangs 58 will prevent rotational movement of the outer races 42, 42'. The bearing cups 32, 36 will also eliminate potential galling of the outer races 42, 42' due to the required axial movement of the shaft 18 and due to sliding friction between the spring 38 and the outer races 42, 42'.

[0032] The description of the invention is merely exemplary in nature
30 and, thus, variations that do not depart from the gist of the invention are intended to be within the scope of the invention. Such variations are not to be regarded as a departure from the spirit and scope of the invention.